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Performance Improvement of R134a Refrigerator Compressor

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PERFORMANCE IMPROVEMENT OF R134a REFRIGERATOR COMPRESSOR

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ABSTRACT

It will bring about many problems for a R12 refrigerator compressor to use R134a as refrigerant directly, such as power increasing, capacity dropping, and noise rising up etc. In order to solve these problems, the mathematical model, optimization model and special test system were built up, many improvements were made by means of the calculating and the testing results. Finally, the performance of R134a refrigerator compressor is greatly improved, much higher than that of the original compressor. Long time operation indicates that the sample compressor works well.

INTRODUCTION

Among all of the current alternative refrigerants of R12, R134a is the most widely used one all over the world. But the differences of thermodynamic and physical properties between R12 and R134a make the performance of a compressor drop down greatly when the R12 compressor directly takes R134a as working medium. Compressor is the main part to consume power in a refrigerator. Therefore, it is very important to improve the performance of the compressor for a R134a refrigerator.

Generally, the performance losses include the dropping of refrigerating capacity, the increasing of energy consumption and the noise rising up. It is necessary to have a deep understanding to the compressor for the performance improvement. A mathematical model, including working processes, valve motions and leakage calculation, is built up in this paper, it will be the base of optimization model. A special testing system which can measure the inner working parameters of hermetic compressor is also built up to determine the key problems and the working directions. The optimization calculation finally defines the concrete improvements.

In this way, many parts, valves, motor, manufacturing and assembling accuracy, muffler, and bearings, etc, were redesigned, the lubricant oil, compressor materials, processing and process materials were changed. At last, the performance of the R134a compressor was greatly improved.

MATHEMATICAL ANALYSIS

1. Working Process Calculation

The first law of thermodynamics is the base of theoretical analysis of any compressors:

$$\frac{dU}{dt}=\frac{\delta Q}{dt}-\frac{\delta W}{dt}+h_i\frac{dm_i}{dt}+\frac{V_i^2}{2}\frac{dm_i}{dt}-h\frac{dm_0}{dt} \tag{1}$$

where, $U=mu$

Q —heat transfer

W —power consumption
h_i —enthalpy of gas input the cylinder
m_i —mass of gas input the cylinder
V_i —flow velocity through the suction valve
h —enthalpy of gas in the cylinder
m_o—mass of gas output the cylinder
t —time
m —mass of gas in the cylinder

It can be used to each process of a compressor. Take the suction process as example, it will be changed:

$$\frac{dT}{dt}=\frac{\left[\frac{V}{m}\left(\frac{\partial P}{\partial v}\right)_T-\left(\frac{\partial h}{\partial v}\right)_T\right]\frac{dV}{dt}+\left\{h_i-h-\left[\frac{V}{m}\left(\frac{\partial P}{\partial v}\right)_T-\left(\frac{\partial h}{\partial v}\right)_T\right]\right\}\frac{dm_i}{dt}}{m\left[\left(\frac{\partial h}{\partial T}\right)_v-\frac{V}{m}\left(\frac{\partial P}{\partial T}\right)_v\right]} \tag{2}$$

where, T —temperature
V —cylinder volume
P —pressure
v —specific volume

Combined with mass equation, motion equation and state equation, the working process of the compressor can be calculated.

2. Valve Calculation

Valve equation is necessary to calculate the displacement of the valve plate, it is also the additional equation of working process simulation.

The energy equations of suction and discharge valves are show as below:

$$h_s\frac{dm_{si}}{dt}-h_{s0}\frac{dm_{s0}}{dt}-\frac{V_{s0}^2}{2}\frac{dm_{s0}}{dt}=0 \tag{3}$$

$$h\frac{dm_{di}}{dt}-h_{d0}\frac{dm_{d0}}{dt}-\frac{V_{d0}^2}{2}\frac{dm_{d0}}{dt}=0 \tag{4}$$

where, the subscript:
d —discharge valve
s —suction valve
i —input the valve
o —output the valve

The motion equation of suction valve is:

$$\rho(x)A(x)\frac{\partial^2 Z}{\partial t^2}=-\frac{\partial^2}{\partial x^2}\left[D\frac{\partial^2 Z}{\partial x^2}\right]+F(x,t) \tag{5}$$

where, ρ —density of valve plate
A —cross section area of valve plate
x —x coordination
Z —Z coordination

D —crooked rigidity of valve plate

F —push force of gas flow

Similarly, to the discharge valve:

$$\frac{\partial^4 Z}{\partial x^4}+\frac{A\rho}{EJ}\frac{\partial^2 Z}{\partial t^2}=\frac{q}{EJ} \tag{6}$$

where, q —force on valve plate

E —modulus of elasticity of valve plate

J —moment of inertia of valve plate section

3. Leakage Calculation

The key problem in leaking calculation is how to determine the flow states in clearance. Based on the following assumptions:

1). oil flow in clearance is laminar flow.

2). no relative motion between the metal surface and the oil layer on the metal surface.

It is obvious that the flow rate of leakage depends on the clearance, the oil viscosity and the pressure difference to be sealed. For a large clearance, the oil entering the clearance has discharged already before it fulls the clearance. In this case, there are oil leakage and gas leakage in the same time. We suppose that the gas flow rate decreases quickly with the decrease of clearance, much faster than the decrease of oil rate. So that there is only oil leakage when the clearance is small enough. Then there must be a critical clearance between oil leakage and oil-gas two-phase leakage.

On the other hand, the flow rate increases with the increase of pressure difference and the decrease of oil viscosity for a certain clearance. Therefore, the oil leakage in a certain clearance at low-pressure difference and high oil viscosity may be changed into oil and gas two-phase leakage at high-pressure difference and low oil viscosity. It means that the critical clearance varies with the working process of the compressor.

From the above-mentioned, critical clearance could be used to distinguish the flow states in the clearance. It is dependent on the oil viscosity and the pressure difference.

Considering the adhesion of oil, we suppose that, for oil gas two phase flow, oil flows along the two metal walls and in the middle of the clearance is the gas flow. The thickness of oil film that can exist on the metal wall is dependent on gas velocity, oil viscosity, pressure difference and wall velocity. Because the gas velocity is only dependent on the state parameters of gas and independent from flow area, the thickness of oil film is also independent on the clearance. So that the maximum thickness of oil film which can exist on the metal wall can be regarded as critical clearance to determine the flow states in the clearance.

It can be thought that there is only oil leakage in the clearance when the total thickness of two oil films is greater than the height of clearance, otherwise, in the clearance is oil and gas two phase flow.

Fig.1 shows the flow model in a clearance. The shearing stress on the boundary surface of oil and gas is:

$$\tau_i=\frac{\lambda_i}{4}\left[\frac{\rho''(V''-V_0)^2}{2}\right] \tag{7}$$

where, τ_i —shearing stress on the boundary surface of oil and gas

λ_i —friction coefficient between oil and gas

ρ'' —density of gas

V'' —mean velocity of gas

V_0 —oil velocity on the boundary surface

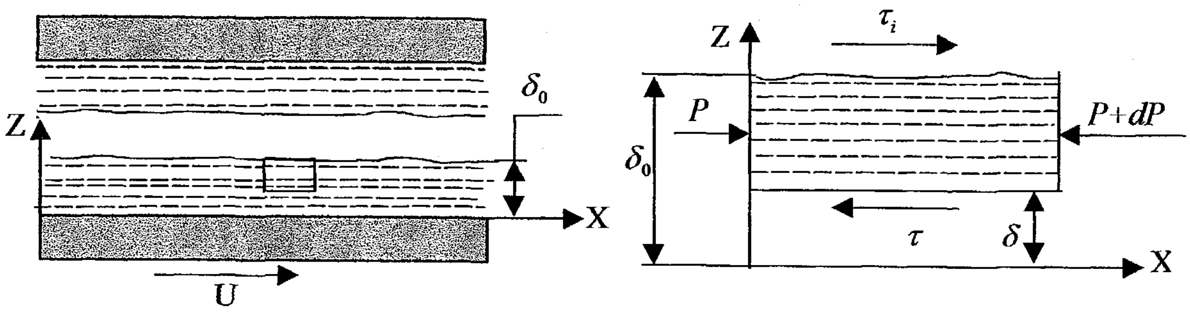

Fig. 1 Flow model in leakage clearance

From Fig.1 the equation of momentum can be built up:

$$\tau_i dx + P(\delta_0 - z) = \tau dx + (P + \frac{dP}{dx}dx)(\delta_0 - z) \quad (8)$$

where, τ —shearing stress between oil layers
δ_0 —thickness of oil film
P —pressure

i.e.

$$\tau = \tau_i - \frac{dP}{dx}(\delta_0 - z) \quad (9)$$

because of:

$$\tau = \mu \frac{dV}{dz} \quad (10)$$

where, μ —viscosity of oil
V —velocity of oil

we obtain the velocity of oil along the oil film:

$$V = \frac{1}{\mu}(\tau_i - \delta_0 \frac{dP}{dx})z + \frac{1}{2\mu}\frac{dP}{dx}z^2 + U \quad (11)$$

where, U—velocity of metal wall

then the oil velocity on the boundary surface is:

$$V_0 = V|_{z=\delta_0}$$
$$= \frac{1}{\mu}\tau_i\delta_0 - \frac{1}{2\mu}\delta_0^{\,2}\frac{dP}{dx} + U \quad (12)$$

and the mean velocity of oil film is:

$$V_m = \frac{1}{\delta_0}\int_0^{\delta_0} V dz$$
$$= \frac{1}{2\mu}\tau_i\delta_0 - \frac{1}{3\mu}\delta_0^{\,2}\frac{dP}{dx} + U \quad (13)$$

On the other hand, the mean velocity of oil film can be calculated from:

$$V_m = V'' / S \quad (14)$$

where, S —sliding ratio between oil and gas

From formulas (7),(12),(13),(14) the thickness of oil film can be determined. The critical clearance is the total thickness of upper and below oil films.

It should be pointed out that the critical clearance varies continuously with the pressure difference and oil viscosity when the compressor works. So that the flow state in the clearance will very while the compressor works, too.

It is easy to calculate the leakage rate whenever the flow state in clearance is determined.

4. Loss Analysis

The theoretical energy consumption is certain for a compressor to compress a certain amount of gas from suction pressure to discharge pressure. Therefore the performance of the compressor is only dependent on the losses.

There are following losses in a hermetic compressor:

1) Motor loss, it depends on the efficiency of the motor.
2) Gas leakage loss. Gas leakage through every leakage path such as valves, the clearance between cylinder and piston will cause the discharge capacity decreasing or the energy consumption increasing.
3) Gas re-expansion loss. It decreases the discharge capacity and increases the gas temperature in suction process.
4) Gas pre-heating loss. Gas heated in suction process makes the discharge capacity decrease and the energy consumption increase.
5) Friction loss. Including bearing friction, friction between cylinder and piston, friction between crank shaft and linkage rod etc.
6) Other losses, such as pressure pulsation loss, valve loss, flow loss etc.

The only way to improve the performance is to reduce these losses. The key problem is that which one is the main loss, i. e. how to obtain the best improvement effect with the lowest cost.

This needs a deep understanding to the compressor, the best way to do it is by means of experiences.

TEST SYSTEM AND RESULTS

Fig. 2 shows the test system specially built up to examine the working conditions of the compressor. It can measure 32 parameters in the meantime, such as every kind of pressures and temperatures, valve displacements, rotating speed, died point, etc.

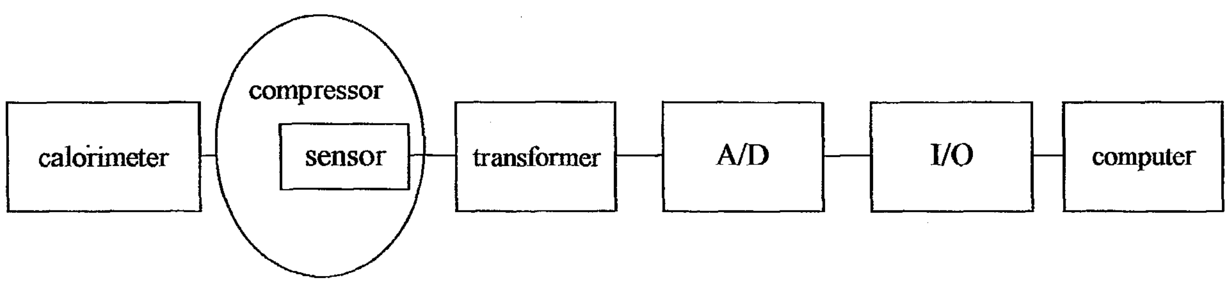

Fig. 2 Test system

The sensors are so mounted that following parameters could be measured:

(1) gas pressure in suction tube
(2) gas temperature in suction tube
(3) surface temperature of suction tube
(4) gas pressure in suction muffler
(5) gas temperature in suction muffler
(6) surface temperature of suction muffler
(7) gas pressure in suction chamber
(8) gas temperature in suction chamber

(9) surface temperature of suction chamber
(10) gas pressure in shell
(11) gas temperature in shell
(12) inner surface temperature of shell
(13) gas pressure in discharge chamber
(14) gas temperature in discharge chamber
(15) surface temperature of discharge chamber
(16) gas pressure in discharge muffler
(17) gas temperature in discharge muffler
(18) surface temperature of discharge muffler
(19) gas pressure in discharge tube
(20) gas temperature in discharge tube
(21) inner surface temperature of discharge tube
(22) surface temperature of cylinder
(23) gas pressure in cylinder

Great amounts of experiences were carried out with this test system. Fig. 3 shows the pressure pulsation in the suction chamber, Fig. 4 and Fig. 5 are the displacements of suction and discharge valves. Table 1 is the test results of gas temperatures and gas pressures. In the figures, H is the valve displacement, θ is the rotating angle of crank-shaft.

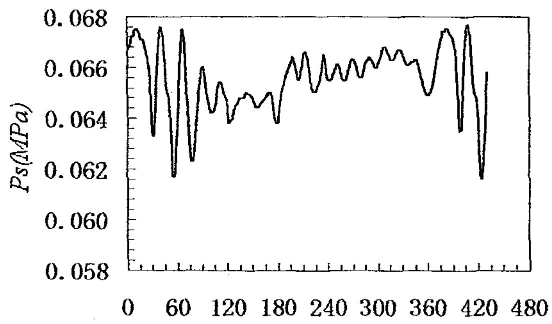

Fig. 3 Pressure pulsation in the suction chamber

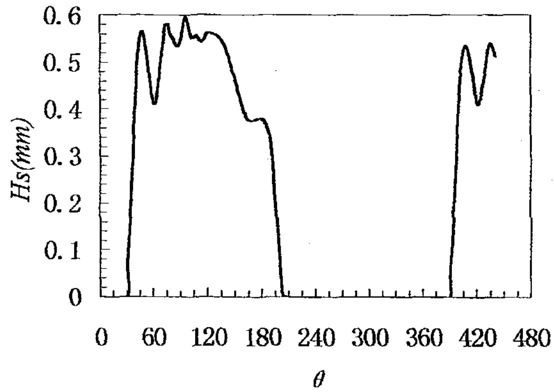

Fig. 4 Displacement of suction valve

Table 1: Gas temperatures and gas pressures

	Temperature (℃)	Pressure (MPa)
Suction tube	24.5	0.1082
Shell	43.3	0.0903
Suction muffler	60.8	0.0712
Suction chamber	87.4	0.0689
Discharge chamber	107.9	1.3874
Discharge muffler	83.0	1.3152
Discharge tube	66.2	1.2136

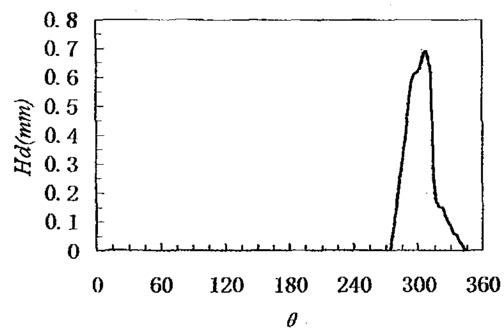

Fig. 5 Displacement of discharge valve

ANALYSIS AND PERFORMANCE IMPROVEMENT

The test results show the inner working conditions of the compressor:

1) There is great pre-heating in the compressor, the gas is heated from 24.5℃ in the suction tube to 87.4℃ in the suction chamber.
2) It exists great pressure pulsation in the suction chamber.
3) Suction valve closes with 23° delay, it means that the spring force of valve plate is too small and will reduce the discharge capacity and valve life.
4) Discharge valve closes with 15° ahead of time, it will cause the trembling motion of the valve plate.
5) There are relative large flow resistance losses.

From mentioned above, it is obvious that the effective way to improve the performance of this compressor is to reduce the suction pre-heating temperature of gas, increase the spring force of suction valve, decrease the spring force of discharge valve, etc.

According to the test results, the compressor was redesigned with optimizing method based on the mathematical model. Meanwhile, other possible steps were taken:

1) Using high efficiency motor.
2) The suction and the discharge muffler systems were redesigned.
3) New belt washers.
4) The valve structure was changed.
5) New lubricant oil and materials.
6) High accuracy of manufacturing and assembling.

In this way, the performance of the compressor was greatly improved. Table 2 compares the performance between improved compressor and original one.

Table 2. Comparison of Performances

	Original compressor	Improved compressor
Capacity (W)	157	152.5
COP (W/W)	1.1	1.32
Noise dB(A)	41	39.6
Vibration (m/s^2)	0.7	0.6

Several thousand-hour life test and long time operation indicate that the sample compressor works well.

CONCLUSION

From all mentioned above, the performance of R134a compressor could be higher than that of R12 compressor. The key problem to improve its performance is correctly determine the main factors, which greatly influence the performance.

The effective way to improve the performance of R134a compressor is to high up the efficiency of the motor, improve the parameters and structure of the valves, reduce the pre-heating and improve the accuracy of manufacturing and assembling.